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Effects of Corrugation Angle on the Performance with Fluid Outlet Positioned above the Plates in Plate Heat Exchangers

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ABSTRACT: The structural design of a heat exchanger plate has a significant impact on the heat exchange efficiency. In this work, the influence of the cold fluid outlet position on the liquid-liquid heat transfer performance of the heat exchanger was numerically investigated. The plate with the cold fluid outlet located in the upper rectangular area has the optimal heat transfer performance. Compared to the typical plate, the heat transfer performance of the optimized structure was improved by 6.2%. Considering both the heat transfer performance and pressure drop characteristics, the comprehensive performance was enhanced by 9.4%. The optimal cold-side corrugation angle is 60°, at which the overall heat transfer coefficient and comprehensive performance coefficient increase by up to 20.9% and 25.8%, respectively. The optimal corrugation angle on the hot fluid side is 22.5° or 30°; the comprehensive performance coefficient is 1.02 times that of the model with a corrugation angle of 15°. The optimal corrugation pitch and height are 9 mm and 1.5 mm, respectively. Meanwhile, altering the fluid inlet position proved to be a strategy for tuning the exchanger's performance characteristics.

KEYWORDS: Plate heat exchanger; plate structure; corrugation angle; numerical simulation

1 Introduction

Plate heat exchangers are widely utilized across thermal management, marine engineering, and industrial energy recovery systems owing to their high thermal efficiency, compact footprint, and operational flexibility [1]. In single-phase liquid-to-liquid heat transfer applications, the thermal-hydraulic performance of plate heat exchangers plays a decisive role in governing overall system efficiency and operating costs. However, conventional plate designs frequently suffer from non-uniform flow distribution across channels, localized fluid stagnation zones, and excessive hydraulic pressure drop, which severely compromise heat transfer effectiveness [2,3]. Therefore, systematically optimizing the plate corrugation geometry and fluid channel layout is essential to eliminate flow maldistribution, minimize hydraulic friction, and improve heat exchange capability.

Recently, simultaneous enhancement of the heat transfer capacity and reduction of the pressure drop has become the primary optimization objective [4]. Karmakar et al. [5] found, through experiments and comprehensive analysis of existing research, that the prediction deviation of performance calculation equations for plate heat exchangers is within an acceptable range. Equation calculations can be used to represent the performance of plate heat exchangers. Jiao et al. [6] employed Computational Fluid Dynamics to perform numerical simulations of air flow and heat transfer inside a louvered finned heat exchanger,

and verified the accuracy of the numerical simulation method through comparison with experimental data. Wasewar et al. [7] used Fluent software to simulate the flow field distribution characteristics inside a plate fin heat exchanger. The results indicated that CFD software can be effectively used for optimizing the design of plate-fin heat exchangers. At the same time, it was found that the flow field distribution in the inter-plate channels has a significant impact on heat transfer performance. Optimizing heat exchangers can be considered from the perspective of flow field distribution. Some researchers have studied the flow characteristics of fluids within herringbone corrugated plates through numerical simulations, suggesting that the fluid flow pattern changes from staggered to tortuous as the corrugation angle increases. This alteration results in a decrease in pressure near the outlet and an increase in pressure between the inlet and outlet due to the modified flow resistance characteristics [8]. Giurgiu et al. [9] proved that a plate with a 60° corrugation angle induced strong turbulence and exhibited better heat transfer efficiency than the plate with a 30° corrugation angle. The overall heat transfer coefficient increased with increasing volumetric flow rate, although the influence of narrow flow channels on pressure drop was not considered. Han et al. [10] established a three-dimensional herringbone corrugated plate heat exchanger model and obtained the velocity, pressure, and temperature cloud images inside the heat exchanger through numerical simulation, indicating that there were heat transfer dead zones and uneven flow phenomena at the inlet and outlet. Nahes et al. [11] proposed a new generalized model for steady-state simulation of gasket plate heat exchangers with single-phase fluids; the average deviation between simulation and experimental was less than 2°C. A novel design scheme for plate heat exchangers was proposed, and experimental investigations were conducted to analyze the heat transfer and pressure drop characteristics within corrugated channels. By fitting the experimental data, empirical relationship formulas for friction factor and heat transfer were derived, revealing that both parameters were significantly influenced by the turbulence level of the fluid. The research highlighted the critical role of inter-plate channel geometry and heat transfer area in optimizing heat exchanger performance [12]. Wang et al. [4] established a two-dimensional physical model of ice storage tanks for smooth and corrugated tube heat exchangers based on natural convection. Due to the corrugation breaking the boundary layer and strengthening fluid mixing, the heat transfer performance was improved by 10.3% when the corrugation spacing was decreased by 41.7%, and improved by 29.4% when the corrugation height was increased by 28%. Dovic et al. [13] studied the flow characteristics in plate heat exchangers by conducting visual tests on channels with corrugation angles of 28° and 61°. When the Reynolds number was in the range of 2 to 10,000, the friction factor and Nusselt number calculated by the proposed correlation were approximately consistent with experimental data. Wang et al. [14] proposed the integration of diversion grooves onto the heat transfer plates to optimize the flow field. The introduction of diversion grooves effectively reduced the fluid distribution non-uniformity coefficient by 6.9%. Concurrently, parametric optimization of the plate geometry demonstrated a substantial impact on thermal-hydraulic performance, yielding a peak improvement of up to 26.2% in the comprehensive coefficient of performance. Although passive turbulators significantly enhance heat transfer efficiency, with reported gains ranging from 20% to 200%, they inevitably introduce hydraulic resistance and elevated pressure drops. Balancing thermal enhancement against pumping penalties using performance evaluation criteria is essential, driving the adoption of high-fidelity computational fluid dynamics simulations and advanced multi-objective optimization strategies, such as artificial intelligence assisted surrogate modeling, to determine optimal geometric parameters [15]. Soomro et al. [16] found that there were significant differences in heat transfer performance among various V-shaped corrugated plate profiles. Cylindrical corrugations exhibited relatively superior heat transfer performance, with a maximum heat transfer rate approximately 35% higher than that of flat plates under the same conditions.

The above studies suggest that the structure of plate heat exchangers has a significant impact on their heat transfer performance, and numerical calculations are an effective way to investigate the performance of plate heat exchangers. In current research, circular holes around the plate are usually set as the inlet and outlet of the fluid [17–19]. While previous studies have focused on the impact of structural deformation at heat exchanger inlets on thermal performance, the core mechanism is that inlet geometric modifications induce changes in fluid channels, thereby altering hydrodynamic characteristics [20,21]. To solve the above problems and optimize the performance of the heat exchanger, this study achieved changes in fluid flow characteristics at the inlet and outlet by adjusting their position and size.

2 Simulation Methods

2.1 Simulation Models

In this study, the heat exchanger model was established based on a simplified plate model of an actual heat exchanger and relevant literature. The three-dimensional model of the heat transfer plate was established based on the dimensional data of the heat exchanger equipment using FEA software. The optimized heat exchanger featured new inlet and outlet positions and sizes in this work. In a typical plate heat exchanger, the four circular holes around the plate are usually set as the fluid inlets and outlets. In this work, the fluid flow direction of the optimized model was as follows: the hot fluid flowed into the plate through the circular hole between the upper-left corner plates and flowed out from the circular hole at the upper-left corner of the plate. The two plate models are shown in Fig. 1; the heat exchanger was composed of alternating stacking of cold- and hot-fluid side plates and gaskets. Due to the change in the gasket of the optimized plate, the cold fluid flowed out from above. The fluid domain between the plates is shown in Fig. 2.

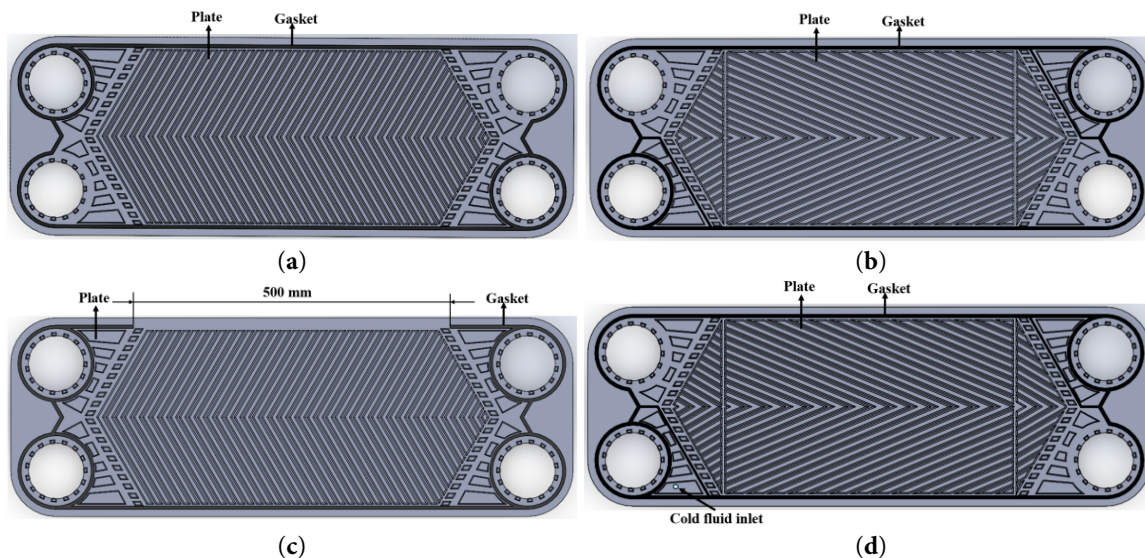


Figure 1: Typical and optimized plate model: (a) Cold-side plate and gasket of typical model; (b) Hot-side plate and gasket of typical model; (c) Cold-side plate and gasket of optimized model; (d) Hot-side plate and gasket of optimized model.

Due to the presence of many small-sized areas in the actual plate, which had little impact on the simulation process of the heat exchanger, certain simplifications were adopted for the areas with less impact in the study. When using CFD software to study the performance of plate heat exchangers, some studies

simplified the model to a certain extent, considering the calculation accuracy and computation time of numerical simulation [22].

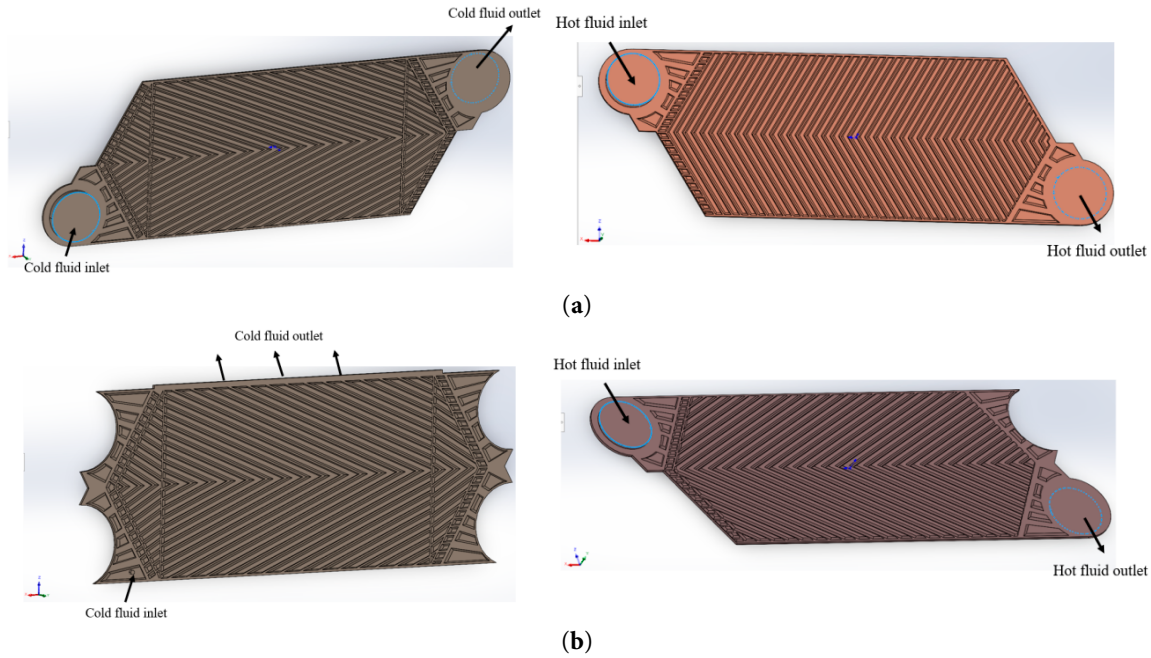


Figure 2: Schematic diagram of fluid inlet and outlet for typical and optimized plate model: (a) fluid domain of typical plates; (b) fluid domain of optimized plates.

The fluid domains between plates were divided into cold and hot fluid domains, and the plates between fluid domains are solid domains. The model was discretized into the model into grids in CFD software, with a minimum size of 1 mm and a maximum size of 2.5 mm. The model body comprises over 4 million grids, distributed across 1.57 million nodes, with a maximum skewness value of 0.598. The software recommends accepting the cell quality when the maximum-skewness is below 0.75. The worst cell quality for this model was 0.13, which exceeds the software's recommended minimum criterion of 0.1. The schematic diagram of the partial model grid is shown in Fig. 3.

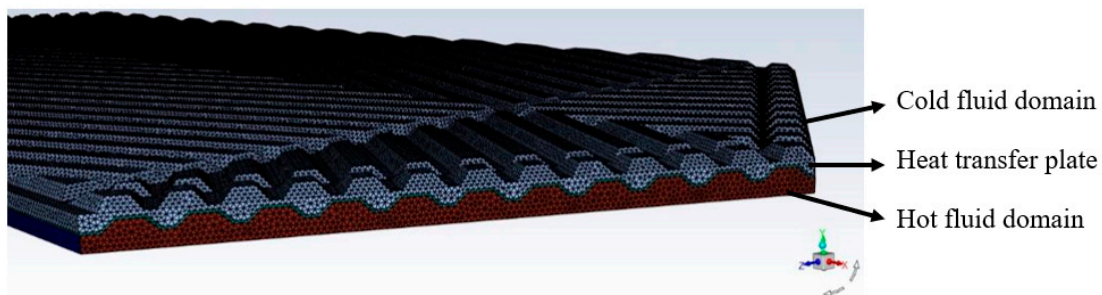


Figure 3: Schematic diagram of the partial model grid.

Previous studies have adopted different turbulence models to perform numerical calculations on the same plate heat exchanger model and compared them with experimental data. The results show that the Realizable $k-\epsilon$ model provides the closest approximation to experimental data and is more suitable for simulating the complex flow inside plate heat exchangers [23]. Therefore, this study adopts the Realizable

$k-\varepsilon$ model for the simulation. While keeping the parameters of the cold fluid region constant, the corrugation angle and pitch of the hot-side plate were systematically varied, as illustrated in Fig. 4.

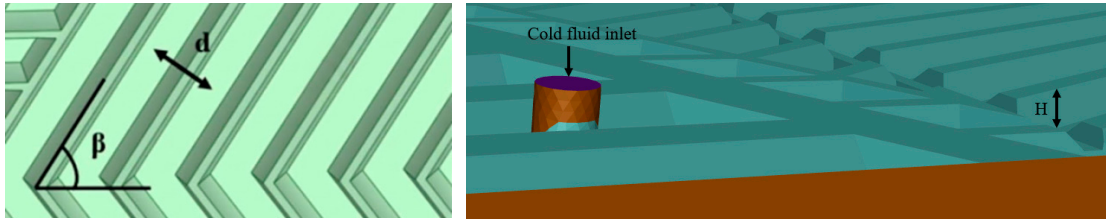


Figure 4: Schematic diagram of corrugation angle (β), corrugation pitch (d) and corrugation height (H) on the model.

2.2 Boundary Condition

Cold water entered the plate passages through a circular orifice with a diameter of 5 mm, while hot water exited through a top rectangular outlet measuring 500 mm $\times$ 3 mm. To study the influence of the corrugation angle on the heat transfer performance of the heat exchanger, the boundary conditions were set as follows: the cold fluid was liquid water with an inlet temperature of 24°C, and the hot fluid was liquid water with an inlet temperature of 100°C. Velocity inlets and pressure outlets were set. Plane 1 is the XZ plane with $Y = 6.42$ mm, which is a suitable cross-section to explore the state of the cold fluid, and plane 2 is the XZ plane with $Y = 2.56$ mm, which is a suitable cross-section to explore the state of the hot fluid. Changes in temperature, velocity, and other parameters across plane 1 and plane 2 were extracted.

In practical applications of plate heat exchangers, the optimal flow velocity inside the inter-plate channels ranges between 0.2 m/s and 0.6 m/s. To strictly regulate channel velocities within this range, extensive preliminary simulations were conducted, yielding prescribed inlet velocities of 7.03 m/s and 0.19 m/s for the cold and hot streams, respectively. Pressure outlet conditions at atmospheric pressure with backflow suppression were assigned at both exits, and an inlet turbulence intensity of 5% was prescribed. No-slip wall conditions were applied to all plate surfaces, with near-wall viscous effects resolved via Enhanced Wall Treatment. The transfer plates were set as a titanium alloy with a thermal conductivity of 16 W/(m·K).

The SIMPLE algorithm was employed for pressure-velocity coupling. A second-order upwind spatial discretization scheme was applied to all transport equations to minimize numerical diffusion and resolve near-corrugation flow patterns and thermal boundary layers. Iterations were continued until scaled energy residuals dropped below 10^{-6} and all remaining governing equation residuals reached below 10^{-5} .

Numerical simulation in this work is based on the assumption that the working medium is an incompressible Newtonian fluid. For water in the temperature range from 24°C to 100°C, the density variation is within 4%. Therefore, the Boussinesq approximation can be adopted, which assumes that the density is treated as constant in all terms of the governing equations except for the gravitational body force term in the momentum equation, where the density is expressed as a linear function of temperature to account for the buoyancy effect.

The fluid control equations are as follows:

- i. Mass flow rate fluids flow equation (http://www.baidu.com/link?url=-_864W8ji1Y7dLwfWeEx86U31PM-XsVW0Ggdk3yoUqtU212GTRaPINOE-xHZzk9A7jQP6DDDkFOlz3LsPoX8dpjuDiU4LjdZM-ga-PgFLuO)

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (1)$$

where, u , v , w is velocity component in X direction, Y direction and Z direction, respectively, the unit is m/s .

ii. Momentum conservation equation

$$\rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial P}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (2)$$

$$\rho \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial P}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \quad (3)$$

$$\rho \left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial P}{\partial z} + \mu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) + \rho \beta (T - T_0) g \quad (4)$$

where ρ is the fluid density, kg/m^3 ; P is the pressure, Pa ; μ is dynamic viscosity, $Pa \cdot s$; β is thermal expansion coefficient; g is gravity in the negative Z -axis direction, m/s^2 .

iii. Energy equation

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad (5)$$

During turbulent flow, u , v , w , P , T represent the instantaneous value of the corresponding physical quantity. α is thermal conductivity, m^2/s .

In the process of fluid motion and heat transfer, the influence of fluid viscosity needs to be considered. The fluid should conform to Newtonian fluid properties, where viscous stress is linearly related to strain rate. Applying the law of conservation of momentum to a moving viscous fluid element yields specific relationships among various flow parameters. The component form of Navier-stokes equation is as follows:

$$\frac{\partial u}{\partial t} = f_x - \frac{1}{\rho} \frac{\partial P}{\partial x} - \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) + \frac{\mu}{\rho} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (6)$$

$$\frac{\partial v}{\partial t} = f_y - \frac{1}{\rho} \frac{\partial P}{\partial y} - \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) + \frac{\mu}{\rho} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \quad (7)$$

$$\frac{\partial w}{\partial t} = g \beta (T - T_0) - \frac{1}{\rho} \frac{\partial P}{\partial z} - \left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) + \frac{\mu}{\rho} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \quad (8)$$

In the N-S equation, P is the static pressure, Pa ; β is the coefficient of thermal expansion of water, $1/K$. f_x , f_y , f_z are the external forces acting on a unit mass of fluid, m/s^2 . Under the conventional setting of no lateral external force and gravity along the negative direction of the Z axis, $f_x = f_y = 0$, $f_z = g \beta (T - T_0)$.

2.3 Data Analysis

The parameters extracted from numerical simulations were processed for data analysis, mainly focusing on the Reynolds number (Re), Nusselt number (Nu), heat transfer factor (j), friction factor (f), comprehensive coefficient of performance (η), and overall heat transfer coefficient (h_t) in plate heat exchangers, which were obtained by the following formulas [14,24,25]:

$$Re = \frac{\rho v d_e}{\mu} \quad (9)$$

$$Nu = \frac{hd_e}{\lambda} \quad (10)$$

$$j = \frac{Nu}{Re \cdot Pr^{1/3}} \quad (11)$$

$$f = \frac{2\Delta P \cdot d_e}{L\rho v^2} \quad (12)$$

$$\eta = \frac{Nu}{f} \quad (13)$$

$$h_t = \frac{1}{\frac{1}{h_1} + \frac{\delta}{\lambda} + \frac{1}{h_2}} \quad (14)$$

In the formula, h is the convective heat transfer coefficient, λ is the thermal conductivity of water, μ is the dynamic viscosity of water, h_1 is the convective heat transfer coefficient on the cold fluid side, h_2 is the convective heat transfer coefficient on the hot fluid side, and δ is the thickness of plate. Reynolds number and Nusselt number are key intermediate quantities for heat transfer performance studying. d_e is the characteristic length, defined here as the hydraulic diameter. It is used to nondimensionalize the governing equations, yielding dimensionless parameters such as Re and Nu to facilitate a unified description of flow and heat transfer across scales. In this study, d_e corresponds to twice the average channel height and is directly extracted from the modeling software.

The overall heat transfer coefficient, which refers to the excellent performance of heat transfer in a series of media, represents the amount of heat exchange in the heat exchanger. The heat transfer factor refers to the intensity of heat transfer inside an object during heat exchange. The friction factor reflects the resistance to fluid flow. The comprehensive performance coefficient considers both heat transfer performance and flow resistance, which is the basis for identifying optimal plate geometries.

2.4 Mesh Independence Verification

To balance computational cost and accuracy, an unstructured tetrahedral mesh was adopted. While insufficient grid resolution leads to under-resolved flow fields, excessive refinement can distort results due to accumulated numerical errors. The mesh independence study, summarized in Fig. 5, confirms that the Nusselt number converges with increasing cell count.

To comprehensively evaluate both the thermal and hydraulic performance of the plate heat exchanger, the average Nu and total channel pressure drop ΔP were selected as the validation criteria for grid independence. Nu reflects the convergence of the thermal boundary layer and temperature field, whereas ΔP is extremely sensitive to the resolution of the boundary layer mesh and vortices in corrugated channels. Utilizing both parameters ensures that both the velocity fields and temperature fields are adequately resolved and independent of grid density.

As shown in Fig. 5a, when the minimum grid size decreased from 1.25 mm to 1 mm and 1 mm to 0.75 mm, Nu decreased by 2.61% and 3.01%, respectively. As shown in Fig. 5b, ΔP is more sensitive to grid size. When the minimum grid size decreased from 1.25 mm to 1 mm and 1 mm to 0.75 mm, ΔP decreased by 2.74% and 3.89%, respectively. When the minimum size of the grid is reduced from 1 mm to 0.75 mm, both key parameters change by less than 3%, which meets the simulation accuracy requirements. But the number of grids increases from 4,157,851 to 8,091,353, resulting in significant improvements in computation

time and computer performance requirements. Therefore, considering all factors, the minimum grid size of 1 mm is selected for subsequent simulation research.

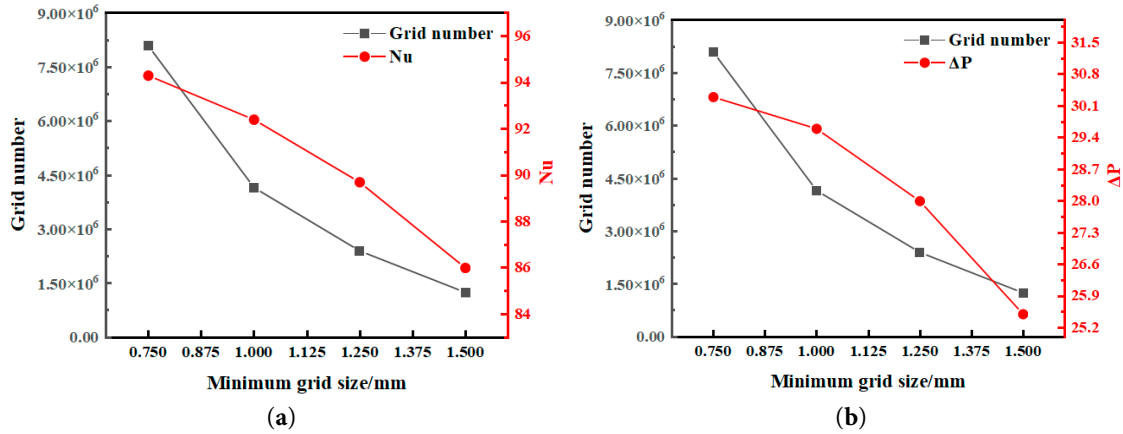


Figure 5: Grid independence analysis for different minimum grid sizes: (a) nusselt number as a function of grid number; (b) pressure drop ΔP as a function of grid number.

This study uses settings with enhanced wall treatment. It was employed in conjunction with the Realizable $k-\epsilon$ model. Enhanced wall treatment seamlessly combines a two-layer model with an enhanced wall function. The corrugated channels of a plate heat exchanger cause frequent fluid separation, reattachment, and strong temperature gradients. When y^+ is small, the heat transfer boundary layer can be accurately resolved. The number of boundary layers was set to 5, with the first-cell height off the wall strictly maintained within 0.2 mm. After checking the post-processing of the simulation, the y^+ of the plates is about 0.97, with maximum local values staying well below $y^+_{\max} < 5$ across the entire Re range. Prism and inflation layers were generated along all solid-fluid heat transfer interface, a transition ratio of 0.3, and a smooth layer growth rate of 1.15.

2.5 Model Comparison Verification

To verify the correctness of simulation assumptions and the reliability of model establishment, traditional plates were simulated under different Re conditions, and the results are shown in Fig. 6.

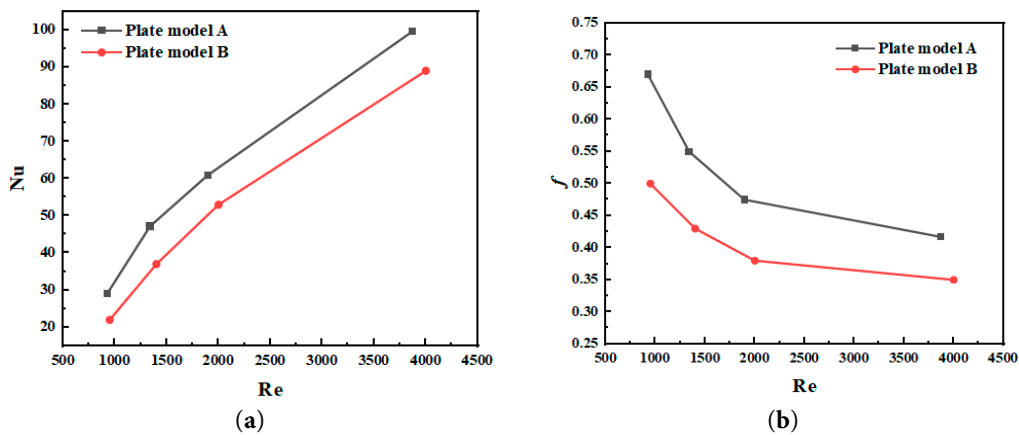


Figure 6: Comparison between this research model and existing research models: (a) the changing trend of Nu; (b) the changing trend of friction factor.

In Fig. 6, the plate model adopts the model established in the previous text, and the simulation method selects Realizable $k-\varepsilon$. The Realizable $k-\varepsilon$ model was specifically selected to predict the turbulent thermal-hydraulic characteristics within the corrugated plate channels. Compared to the Standard $k-\varepsilon$ model, the Realizable $k-\varepsilon$ model combines a formulation for turbulent viscosity with a dissipation rate transport equation based on the equations of vortex flow dynamics. In corrugated plate heat exchangers, the fluid experiences continuous redirection and pronounced boundary layer separation. The Realizable $k-\varepsilon$ model offers superior fidelity in capturing these complex flow physics, accurately predicting local boundary layer detachment, reattachment, and the associated convective heat transfer rates while maintaining robust numerical stability and computational efficiency.

Plate model A was the traditional heat transfer plate model established in this study, while Plate model B was the plate model previously studied [26]. By controlling the fluid inlet flow rate, the Re was kept within the same range. As shown in Fig. 6, when the Re varied, the trends of Nu and f for the two models were the same. When the Re increased from approximately 1400 to approximately 2000, the Nu of the two models increased by 36.0% and 43.1%, respectively. When the Re increased from approximately 2000 to approximately 4000, the Nu of the two models increased by 63.5% and 67.9%, respectively. When the Re increased from around 1400 to around 2000, the friction factor of the two models decreased by 11.6% and 12.7%, respectively. When the Re increased from approximately 2000 to approximately 4000, the friction factor of the two models decreased by 8.1% and 12.2%, respectively.

Due to partial differences in the models and boundary conditions of the two types of plates, there are differences in the numerical values of the two parameters when the Re are close. However, the changing trends of Nu and f are basically the same, indicating that the simulation method used in this study is acceptable.

3 Result and Discussion

3.1 Influence of Fluid Inlet and Outlet Position

Previous research indicates that the optimal corrugation angle for traditional plate heat exchangers is typically 60° . To investigate whether the performance of the newly proposed plate model could be improved, this study held other structural and boundary parameters constant to solely modify the inlet and outlet configurations, thereby maintaining a consistent corrugation angle of 60° for simulation and performance comparison. Two types of models were established: the optimized plate, with a small circular inlet at the lower-left corner and a top-boundary outlet directly above, and the typical plate, with a standard circular outlet at the upper-right corner. To simulate the dimensional parameters of heat exchange plates in practical applications, the corrugation pitch was set to 9 mm and the corrugation height to 1.5 mm.

As shown in Figs. 7 and 8, the cold fluid flowed into the inter-plate channel from the lower left corner, and its temperature gradually increased in the flow channel. The hot fluid flowed into the inter-plate channel from the upper left corner, and its temperature gradually decreased in the flow channel. When the outlet was directly above in Fig. 8, a significant difference was observed in the distribution of cold fluid on the outlet cross-section, indicating that the flow distribution was more reasonable. When the outlet was located at the circular hole area in the upper right corner in Fig. 7, the flow rate and velocity at the cold fluid outlet increased significantly, inducing an increase in pressure.

Due to the closer distance between the outlet and the inlet of the cold fluid in the optimized plate, the fluid flow resistance was reduced compared to that in the typical plate. Meanwhile, as the cross-section of the cold fluid inlet was reduced in the optimized plate, the velocity increased significantly, thereby enhancing the turbulence is significantly increased and the heat transfer effect relative to the original plate. In addition,

when fluid flows in the grooves between plates, it is subjected to tangential forces from the relative plates. The tangential force is perpendicular to the direction of fluid flow, inducing vortical flow. The upper area of the typical plate is closed; the fluid flows along the groove on one side of the plate. When it reaches the edge of the plate, it is reflected and deflected to the remaining area between plates, resulting in a cross-shaped flow [27]. In the upper area of the optimized plate, flow reflection is eliminated, and vortices are generated when a certain corrugation angle is reached, which can improve the heat transfer efficiency.

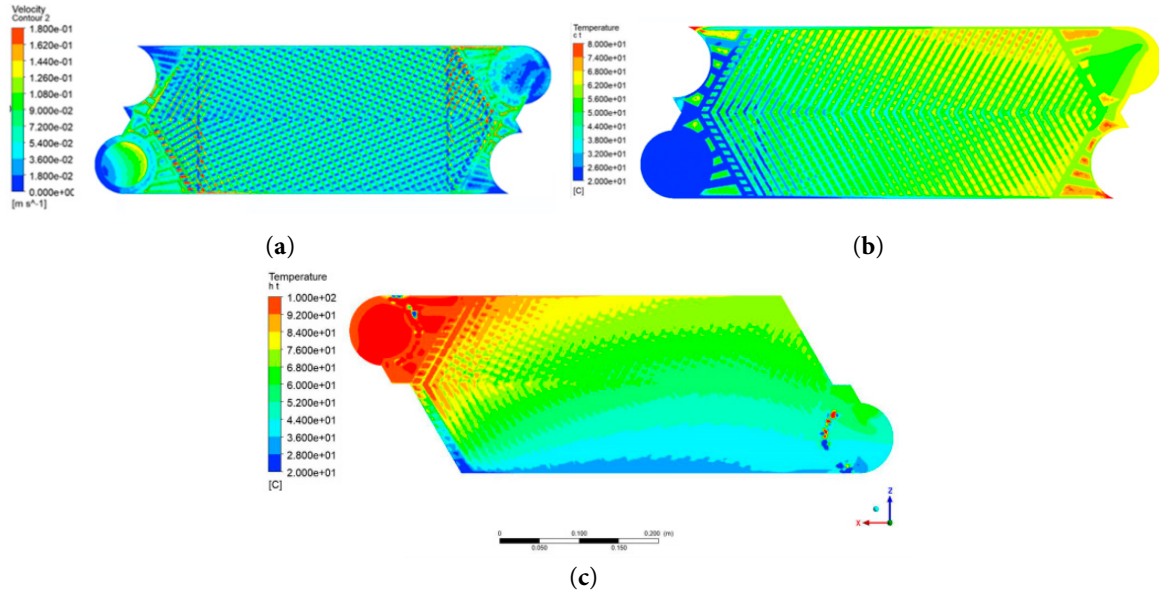


Figure 7: The temperature and velocity distribution in typical plate: (a) temperature distribution in cold fluid domain; (b) velocity distribution in cold fluid domain; (c) temperature distribution in hot fluid domain.

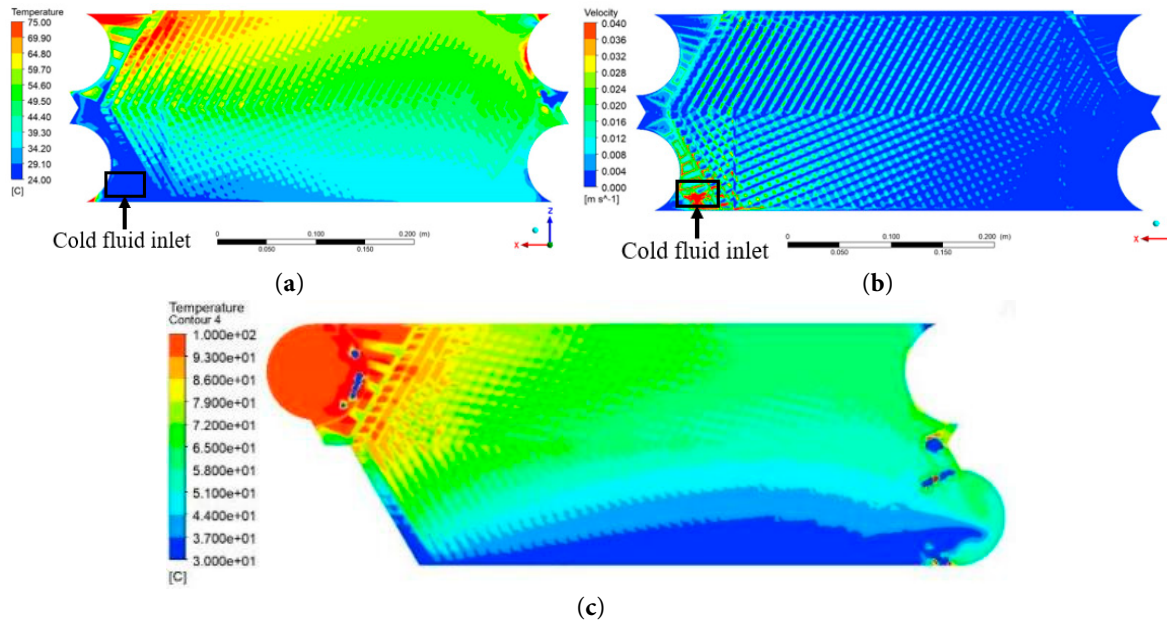


Figure 8: The temperature and velocity distribution in optimized plate: (a) temperature distribution in cold fluid domain; (b) velocity distribution in cold fluid domain; (c) temperature distribution in hot fluid domain.

Export the key thermodynamic parameters of the two types of plates through post-processing and calculate them. The calculation results are shown in Table 1. As shown in Table 1, compared to the typical plate, the convective heat transfer coefficient was increased by 6.8%, and the friction factor was decreased by 7.0% on the cold fluid side in the optimized plate. As for the hot fluid side, the convective heat transfer coefficient was increased by 6.3%, and the friction factor was decreased by 5.6%.

Table 1: Thermodynamic properties and comprehensive performance of typical and optimized plates.

Plates	Typical Plate	Optimized Plate
Convective heat transfer coefficient at cold fluid side ($\text{W}/\text{m}^2\cdot\text{K}$)	4146.31	4428.26
Convective heat transfer coefficient at hot fluid side ($\text{W}/\text{m}^2\cdot\text{K}$)	3867.02	4109.97
Overall heat transfer coefficient ($\text{W}/\text{m}^2\cdot\text{K}$)	1768.79	1870.03
Friction factor at cold fluid side	2.13	1.98
Friction factor at hot fluid side	1.96	1.85
Comprehensive coefficient of performance	61.32	67.08

The overall heat transfer coefficient rose from $1768.79 \text{ W}/(\text{m}^2\cdot\text{K})$ to $1870.03 \text{ W}/(\text{m}^2\cdot\text{K})$, representing a 5.7% increase. The optimized plate underwent changes in fluid flow area, and the cold fluid to reach the outlet faster, which reduced the resistance of the fluid on the plate. Meanwhile, due to the reduction of the inlet cross-section, fluid velocity increasing of the optimized plate leads to an increase in fluid turbulence, promoting heat exchange. Considering both the heat transfer performance and resistance characteristics, the comprehensive performance coefficient of the optimized plate was improved by 9.4% compared to the typical plate. It suggests that the change of the inlet and outlet configurations has a positive effect on the performance of the plate heat exchanger.

3.2 Influence of Corrugation Angle at Cold Fluid Side

The corrugation angle is critical to the performance of the plate. In Section 3.1, it was demonstrated that the performance of the optimized plate, in which the cold fluid outlet is directly above, was better than that of the typical plate. Based on this plate, the influence of the corrugation angle on the optimized heat exchanger was further studied.

According to existing research on the performance of plate heat exchangers, the optimal corrugation angle is usually around 60° . The shape of channels between the plates has a significant impact on the fluid flow state, and the corrugation angle directly affects the heat transfer performance of the heat exchanger. To obtain the optimal corrugation angle of the optimized plate, plate models with corrugation angles of 15° , 30° , 45° , 60° , and 75° at the cold fluid side were set when the corrugation angle remaining at 60° at the hot fluid side.

As shown in Fig. 9, when the corrugation angle of the plate on the cold fluid side increased from 15° to 60° , the average temperature in the upper left corner of the cold fluid domain gradually increased and the heat transfer performance in the fluid domain gradually improved. When the corrugation angle of the plate on the cold fluid side increased from 60° to 75° , the temperature change in the fluid domain decreased, indicating that the heat transfer effect decreased in this corrugation angle range. In Fig. 10, as the corrugation angle of the plate on the cold fluid side increases, the velocity difference in the cold fluid domain increased. When the corrugation angle of the plate on the cold fluid side increased to 60° , the fluid flow still exhibited a cross flow. At 75° , the cross flow disappeared. As the cross flow reflects the degree

of fluid contact points, the heat transfer performance of the plate with a corrugation angle of 75° was not significantly improved. The increase in the corrugation angle will also lead to an increase in the fluid flow resistance, and it is necessary to comprehensively consider the heat transfer performance and resistance characteristics to select the optimal corrugation angle. The results indicate that the optimal corrugation angle on the cold fluid side was 60° . Previous studies have analyzed the influence of corrugations on fluid heat transfer and also confirmed that a plate with a corrugation angle of 60° was the best plate for heat transfer.

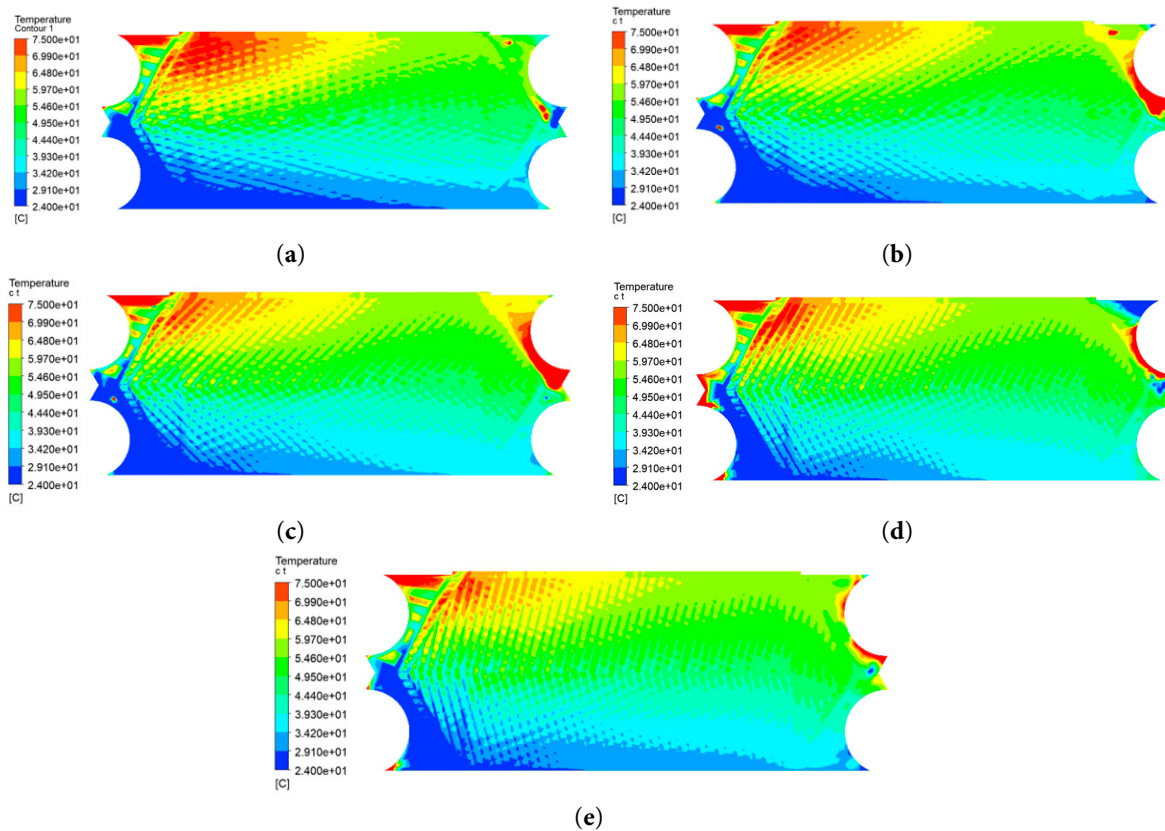


Figure 9: Temperature variations of the fluid on Plane 1 (XZ plane with $Y = 6.42$ mm) with different corrugation angle of cold side: (a) 15° ; (b) 30° ; (c). 45° ; (d) 60° ; (e). 75° .

To quantitatively compare the performance of different corrugation angles, the parameters of the plate with a corrugation angle of 15° were set to 1.0 as the baseline against the results of other angles. The results are shown in Fig. 11; the convective heat transfer coefficients on the cold fluid side are increased by 6.1%, 18.9%, 29.1%, and 27.2% under the corrugation angles of 30° , 45° , 60° , and 75° , respectively. The overall heat transfer coefficient increased by 5.0%, 13.1%, 20.9%, and 21.8%, respectively. Due to the increase in the corrugation angle, the flow state of the fluid between the plates gradually developed from a relatively gentle flow to a turbulent flow. Compared with the corrugation angle of 15° , the friction factor on the cold fluid side decreased by 8.1% and 6.0% when the corrugation angles were 30° and 45° , respectively. However, the friction factor on the cold fluid side increased by 1.5% and 10.9% when the corrugation angles were 60° and 75° , respectively. As the corrugation angle increases, the fluid will be obstructed by the corrugation in the fluid channel, and the tangential force component will become opposite to the flow direction [28]. As the corrugation angle increased, the pressure drop gradually increased [29].

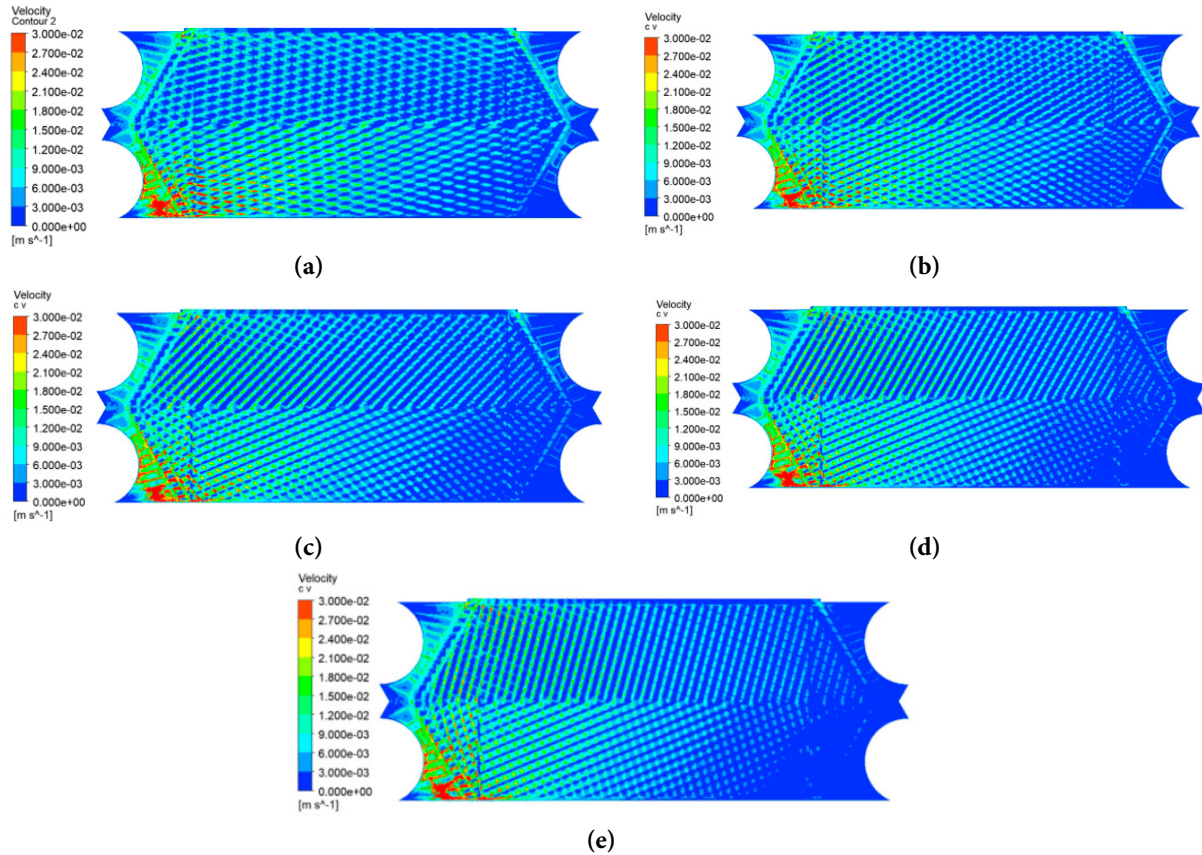


Figure 10: Velocity variations of the fluid on Plane 1 (XZ plane with $Y = 6.42$ mm) with different corrugation angle of cold side: (a) 15° ; (b) 30° ; (c) 45° ; (d) 60° ; (e) 75° .

Since the comprehensive performance coefficient both considers the heat transfer performance and resistance characteristics, it is generally believed that the higher the comprehensive coefficient of performance, the better the overall plate performance. According to the results in Fig. 11c, when the corrugation angle on the cold fluid side was 60° , the comprehensive performance coefficient increased by 25.8% compared to the model with a 15° corrugation angle. The comprehensive performance of the plate was best when the corrugation angle was approximately 60° , which is the trend of plate changes in previous work [30,31]. As the corrugation angle increased, the heat transfer coefficient gradually increased, and the resistance rose sharply, so it is not necessary that plates with higher heat transfer coefficients are better. This is because the fluid flow between the plates gradually changed from a simple two-dimensional flow to cross flow as the corrugation angle increased. The number of contacts between the fluid and plates increasing rendered heat transfer more complete. Additionally, due to the flow disorder degree increases, the resistance of the fluid increases during the flow process, causing the comprehensive performance coefficient to exhibit an initial rise followed by a decline.

3.3 Influence of Corrugation Angle at Hot Fluid Side

Based on the results obtained in Section 3.2, compared to the plate with a corrugation angle of 15° , the convective heat transfer coefficients of the plate with a corrugation angle of 60° on the cold fluid side increased by 29.1%, the overall heat transfer coefficient increased by 20.9%, whereas the friction factor on the cold fluid side only increased by 1.5%. This comprehensive comparison confirms that 60° is the optimal

cold-side corrugation angle. On the basis of these findings, models of plates with corrugation angles of 10° , 22.5° , 35° , 45° , 60° , and 75° at hot fluid side were set up.

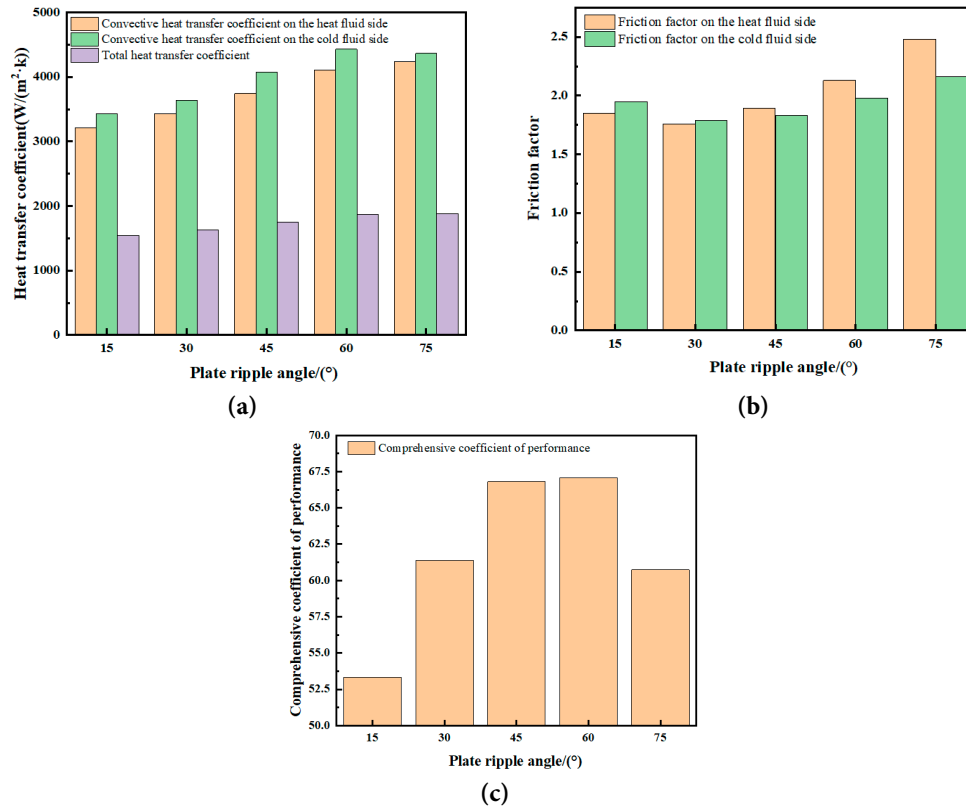


Figure 11: Influence of corrugation angle at cold fluid side on plate performance: (a) heat transfer coefficients; (b) friction factor; (c) coefficient of performance.

The temperature distribution as shown in Fig. 12, suggesting that the temperature of the cold fluid gradually increased from the inlet in the lower left corner to the outlet in the upper right corner. When the corrugation angle on the hot fluid side increased from 10° to approximately 45° , the average temperature in the upper left corner of the cold fluid domain gradually increased, and the temperature change in the entire fluid domain increased. When the corrugation angle at the hot fluid side increased from 45° to 75° , the temperature change in the fluid domain decreased. The heat exchange between the cold and hot fluid domains is most effective when the corrugation angle on the hot fluid side is around 10° to 45° .

To quantitatively compare the performance of different corrugation angles, the parameters of plate with a corrugation angle of 10° were set to 1.0 as the baseline. In Fig. 13, when the corrugation angle on the hot fluid side gradually increased from 10° to 22.5° , 35° , 45° , 60° , and 75° , the convective heat transfer coefficients increased by 10.9%, 12.2%, 11.4%, 6.1%, and 2.2%, respectively. The overall heat transfer coefficient increased by 8.5%, 8.8%, 7.2%, 5.1%, and 1.6%, respectively. As the corrugation angle increased, the movement time of the fluid between the plates also increased, which increased the heat transfer.

When the corrugation angle increased, the obstruction of fluid flow increased, and some fluid moved along the corrugation to the lower area by gravity. The fluid mass in the middle of the plate, which is the main heat transfer area, decreased, and the heat transfer capacity became weaker. When the corrugation angle gradually increased from 10° to 22.5° , 35° , 45° , 60° , and 75° , the friction factor increased by 1.3%, 6.2%,

13.6%, 23.9%, and 39.7%. Based on the comprehensive performance coefficient, the optimal corrugation angle on the hot fluid side was found to be within 22.5° to 35° .

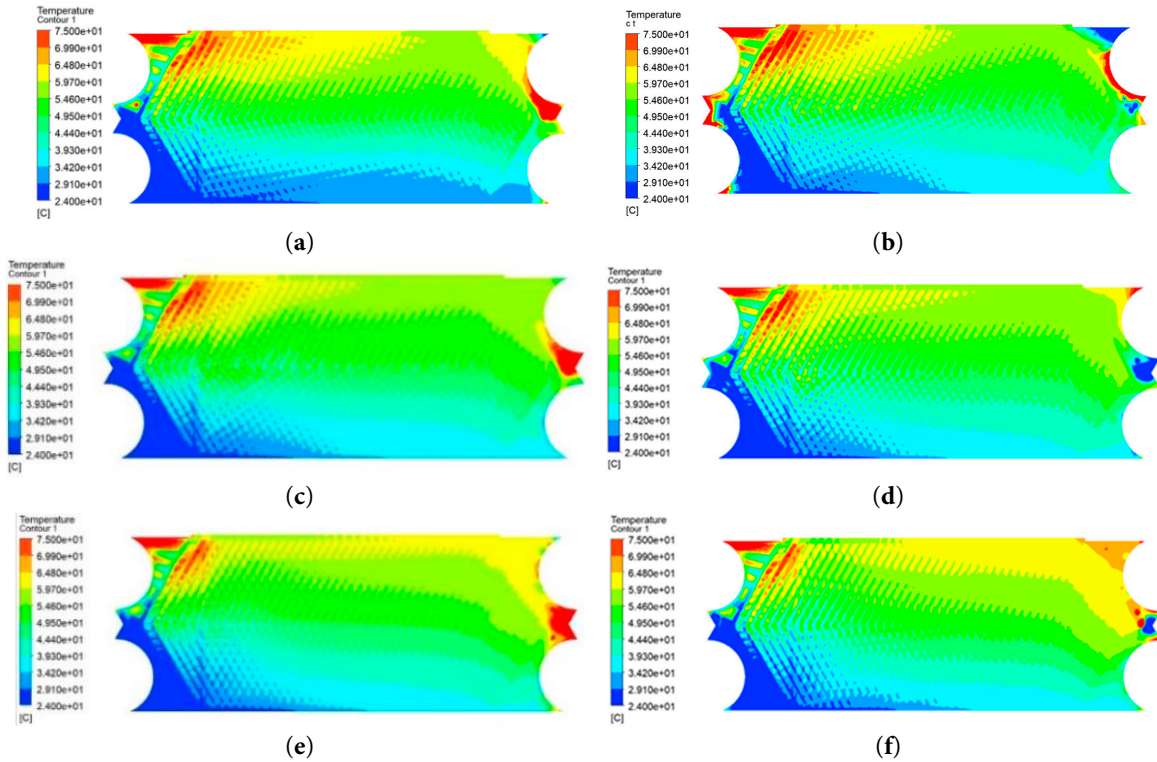


Figure 12: Temperature variations of the fluid on Plane 1 with different corrugation angle of hot side: (a) 10° ; (b) 22.5° ; (c). 35° ; (d) 45° ; (e). 60° ; (f). 75° .

According to existing research [32], because a component of the tangential force is opposite to the mainstream direction, the pressure gradient increases as the plate corrugation angle increases, resulting in a higher pressure drop. The increase in corrugation angle causes the fluid flow to transform from a tortuous flow to a complex vortex flow. Turbulence is the fundamental cause for enhancing in plate heat exchangers, and intensification of turbulence improves the heat transfer effect. In addition, the interaction between vortex structures inside the fluid leads to energy dissipation and an increase in resistance. The results in Fig. 13 indicate that as the corrugation angle on the hot fluid side gradually increased, the heat transfer performance of the plate heat exchanger exhibited an initial increase followed by a decrease, and the resistance characteristics gradually increased. When the corrugation angle at the hot fluid side was 22.5° or 35° , the comprehensive performance coefficient was more than 1.02 times that of 15° . When the corrugation angle at the hot fluid side increased from 45° to 75° , the comprehensive performance coefficient gradually decreased. Therefore, considering the comprehensive coefficient of performance, the optimal corrugation angle at the hot fluid side ranges from 22.5° to 35° .

3.4 Influence of Corrugation Pitch and Height

Based on the foregoing analysis, the plate configuration with a corrugation angle of 60° on the cold side and 22.5° on the hot side exhibited the optimal overall thermal-hydraulic performance. Existing literature indicates that the corrugation pitch and height govern the flow channel geometry and, consequently, the turbulence intensity during heat exchange. In the developed models, the corrugation width was fixed at

approximately 6 mm, where the pitch is defined as the sum of the corrugation width and the intermediate channel width. Adhering to the single-variable principle, a parametric study was conducted by varying the corrugation pitch on the cold to 7.5, 9, 10.5, 12, and 13.5 mm to evaluate its influence. Similarly, the corrugation height was varied independently at 1.2, 1.5, 1.8, 2.1, and 2.4 mm. The key performance parameters corresponding to these geometries are summarized in Tables 2 and 3, respectively.

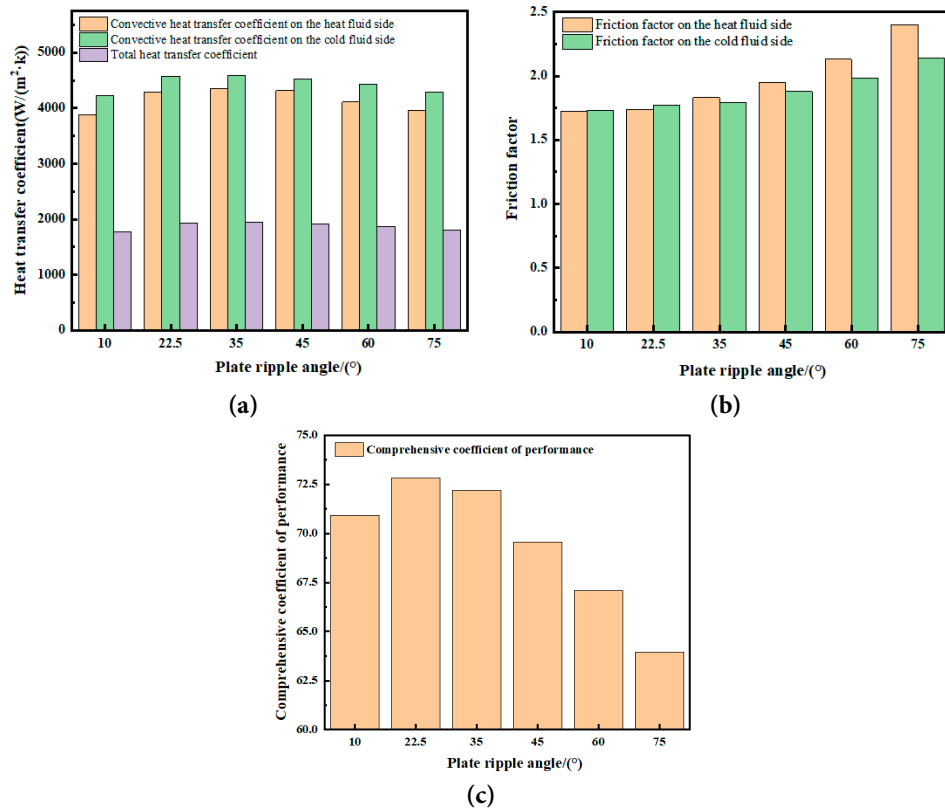


Figure 13: Influence of corrugation angle at hot fluid side on plate performance: (a) heat transfer coefficients; (b) friction factor; (c) coefficient of performance.

Table 2: Performance table of plates with different corrugation pitch.

Different corrugation pitch (mm)	7.5	9	10.5	12	13.5
Heat transfer coefficients (W/m ² ·K)	1962.5	1870.0	1611.7	1526.5	1435.1
Cold fluid side friction factor	2.23	1.98	1.75	1.64	1.57
Comprehensive coefficient of performance	66.93	67.08	65.75	63.93	61.59

Table 3: Performance table of plates with different corrugation height.

Different corrugation height (mm)	1.2	1.5	1.8	2.1	2.4
Heat transfer coefficients (W/m ² ·K)	1721.9	1870.0	1966.5	2041.2	2210.6
Cold fluid side friction factor	1.73	1.98	2.21	2.54	2.91
Comprehensive coefficient of performance	63.12	67.08	67.16	64.25	62.83

The performance of plates with different corrugation pitches as shown in Table 2. With an increasing corrugation pitch, the overall heat transfer coefficient decreased monotonically from 1962.5 to 1435.1 W/(m²·K), corresponding to a 26.9% reduction. The friction factor also exhibited a monotonic decline, dropping by 29.6% from 2.23 to 1.57. In contrast, the comprehensive coefficient of performance of the cold side followed a parabolic trend, initially rising to a peak value of 67.08 at a pitch of 9 mm before decreasing. This peak represents an improvement of up to 8.9% over the other configurations.

The influence of corrugation pitch on intra-channel flow is primarily manifested through modifications to the velocity distribution and turbulence intensity. The cross-sectional flow area governs the hydraulic diameter, which directly dictates the Reynolds number. As the fluid traverses the corrugated structure, the boundary layer separates at the troughs and reattaches at the crests; the pitch determines the streamwise distance between separation and reattachment. At smaller pitches, the boundary layer enters the subsequent corrugation cycle before becoming fully developed, thereby increasing the frequency of separation-reattachment events and leading to a less uniform flow field distribution.

The performance of plates with different corrugation heights is shown in Table 3. With increasing corrugation height, the overall heat transfer coefficient increased monotonically from 1721.9 to 2210.6 W/(m²·K), representing a rise of 28.4%. The friction factor showed a similar monotonic increase of 68.2%, climbing from 1.73 to 2.91. The comprehensive coefficient of performance of the cold side variation with height also exhibited a non-monotonic trend, peaking at 67.16 at a height of 1.8 mm. As shown in Table 3, when the corrugation height increased from 1.5 mm to 1.8 mm, the comprehensive coefficient of performance enhancement was negligible, whereas the flow resistance experienced a notable increase of approximately 15.1%. Consequently, considering both heat transfer augmentation and the pressure penalty, a corrugation height of 1.5 mm was determined as the optimal dimension for subsequent numerical studies.

Corrugation height significantly influences the thermal-hydraulic performance by altering the geometric confinement and hydrodynamic characteristics of the flow channel, albeit with differential sensitivities for heat transfer and pressure drop. The enhancement in heat transfer relies primarily on the intensified turbulence and improved utilization of the effective heat transfer area. As the corrugation height increases, the streamwise undulation of the channel becomes more pronounced, reducing the minimum flow area at the troughs while increasing the turning radius. This amplifies local acceleration during flow redirection and elevates the risk of boundary layer separation. Concurrently, the elevated flow velocities result in higher local form drag and augmented frictional losses along the channel. Consequently, both the heat transfer coefficient and pressure drop exhibit a monotonic increasing trend with rising corrugation height.

3.5 Influence of Cold Fluid Inlet Position

As established in the preceding analysis, the optimal plate configuration (Plate 1) after modifying the inlet and outlet ports features a cold and hot side corrugation angle of 60° and 22.5°, a corrugation pitch of 9 mm, and a height of 1.5 mm. Based on the model, the cold fluid inlet position was parametrically varied to investigate its impact on the thermal-hydraulic performance. In the above model, cold water enters the plate passages through a circular orifice with a diameter of 5 mm and exits through a rectangular top outlet measuring 500 mm × 3 mm.

The resulting modified plate geometries (Plates 2 and 3) are illustrated in Fig. 14. The hot water outlet in Plate 2 and Plate 3 is identical to that of the Plate 1, featuring a top rectangular opening of 500 mm × 3 mm. The cold fluid inlets consist of circular holes with a diameter of 5 mm. Compared with Plate 1, Plate 2 shifts the inlet position upward, while Plate 3 incorporates the cold fluid inlets from both previous configurations. To ensure a fair comparison of thermal performance, a constant total volumetric flow rate was maintained

across all plates, the flow rate at each individual inlet of Plate 3 was reduced to 50% of that used in the other two plate configurations.

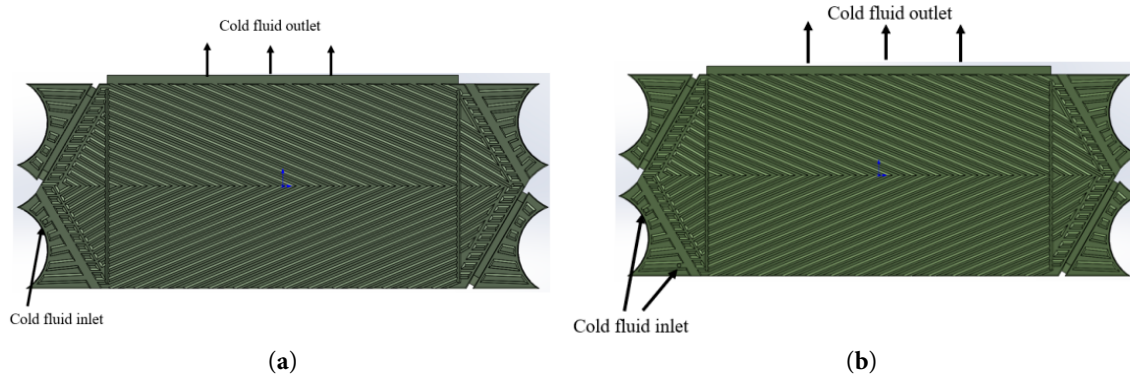


Figure 14: Change the rear plate model of the cold fluid inlet: (a) plate gasket model 2 (Changing the position of the cold fluid inlet); (b) plate model 3 (Adding inlet without changing flow rate).

The temperature distribution cloud map of the cold fluid outlet section was obtained by simulating three models, as shown in Fig. 15. In Fig. 15, the cold-fluid outlet velocity follows a non-monotonic trend—initially increasing before declining with flow distance—for all models. Notably, Plates 2 and 3 achieved higher flow velocities, with Plate 2 reaching the maximum. Specifically, relocating the inlet upward intensified the flow in the upper-left region of Plate 2. Elevated velocities were consistently observed near the hot-fluid inlet across all models, corresponding to peak local flow rates. This localized acceleration induced a marked temperature drop adjacent to the inlet, followed by a gradual velocity decay toward the outlet.

The fluid flow rate input to each plate is the same, and the plate structure is similar, with only a change in the position of the cold fluid inlet. Structural analysis further elucidates these performance variations. Plate 1 comprises protruding diversion distribution and herringbone corrugations. Since Plates 2 and 3 preserve this core architecture while modifying only the vicinity of the inlet to alter local flow dynamics, the performance differentials among the three designs remain relatively marginal. Table 4 presents the main thermodynamic parameters of the three models under the same input conditions.

Table 4: Simulation results for plate heat exchangers with different geometries.

Plate Model	1	2	3
Convective heat transfer coefficient at cold fluid side ($\text{W}/\text{m}^2\cdot\text{K}$)	4428.26	4507.32	4295.40
Convective heat transfer coefficient at hot fluid side ($\text{W}/\text{m}^2\cdot\text{K}$)	4109.97	4093.53	4031.89
Overall heat transfer coefficient ($\text{W}/\text{m}^2\cdot\text{K}$)	1870.03	1926.13	1836.37
Friction factor at cold fluid side	1.98	2.08	2.05
Friction factor at hot fluid side	1.85	1.81	1.78
Comprehensive coefficient of performance	67.08	68.10	66.41

Simulation results reveal distinct thermal-hydraulic characteristics among the three plate configurations. Plates 2 and 3 exhibit a more uniform temperature distribution across the outlet plane, with relatively higher temperatures observed near the cold-fluid inlet. In contrast, as established in the prior analysis for Plate 1, the cold-side temperature is elevated adjacent to the hot-fluid inlet, a phenomenon attributable to the underlying velocity distribution profile. Plate 2 achieves the overall heat transfer coefficient of

1926.13 W/(m²·K), representing an improvement of 3.0% and 4.9% compared with the other two configurations. Furthermore, it exhibits the highest comprehensive coefficient of performance, surpassing the alternatives by 1.5% to 2.5%.

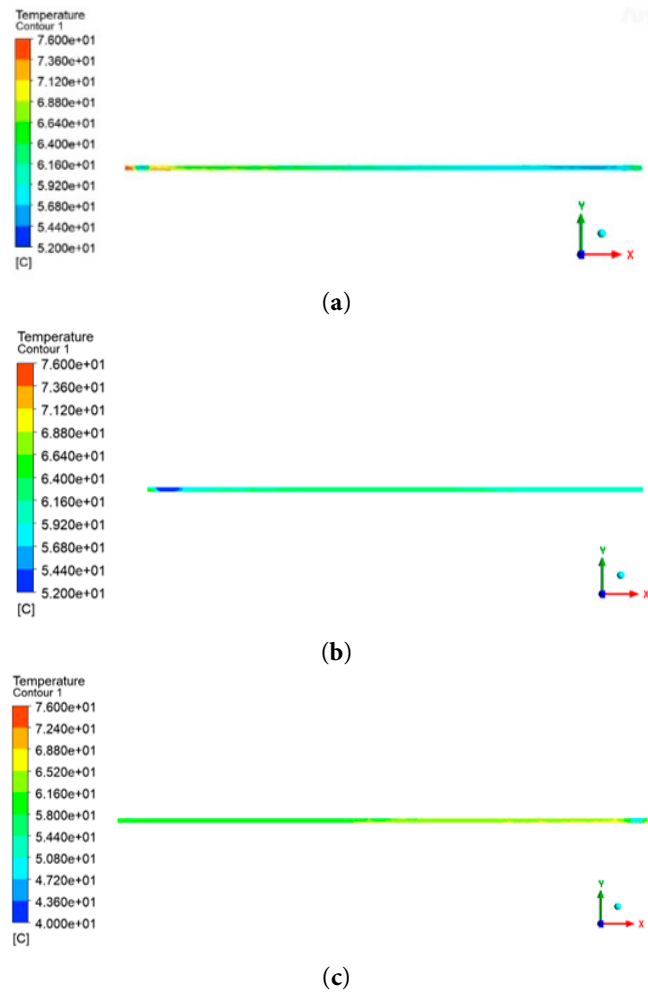


Figure 15: Temperature distribution of cold fluid outlet section: (a) plate model 1; (b) plate model 2; (c) plate model 3.

Fig. 16 presents the velocity vector diagrams of the XZ and YZ cross-sections at the inlet. Due to the narrow cross-section of the inlet, the velocity increases significantly near the inlet. As the distance from the fluid inlet increases, the cross-sectional area of the channel increases, and the velocity gradually decreases under a constant flow rate.

Fig. 16 also reflects the guiding and hindering effects of the plate structure on the fluid flow process. The plate structure induces flow resistance, extending the fluid residence time within the inter-plate heat transfer channels. At the same time, the fluid induces recirculation on the plate, which impacts the wall and disrupts the thermal boundary layer, thereby improving heat transfer performance. However, the structural flow obstruction will significantly enhance the flow resistance, resulting in a decrease in the actual mass flow rate in the system circuit. It is necessary to comprehensively evaluate the effects of heat transfer enhancement and flow resistance on the heat exchanger. Consequently, considering both thermal enhancement and pressure penalty, Plate 2 proves to be the optimal design among the three configurations.

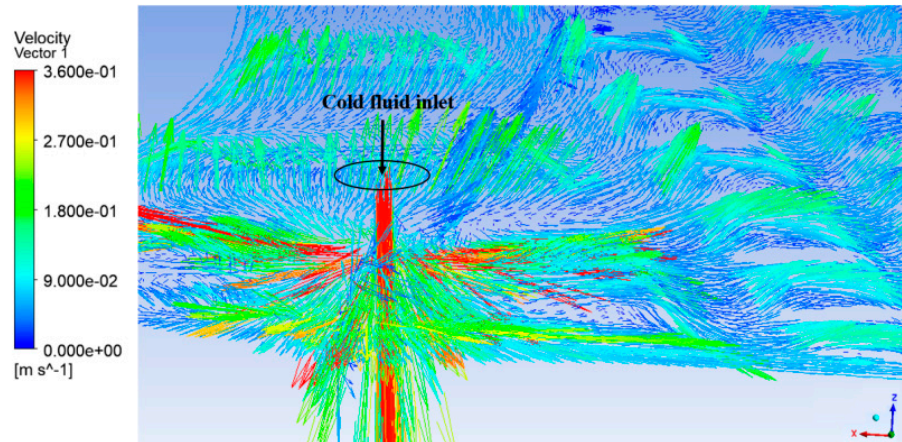


Figure 16: Cross section velocity vector diagram at the inlet.

4 Conclusion

This study proposed a novel plate heat exchanger featuring a top-mounted rectangular outlet, which demonstrates superior performance over the conventional circular-hole design, achieving a 9.4% improvement in the comprehensive coefficient of performance. Parametric analysis revealed that a cold-side corrugation angle of 60° maximized thermal-hydraulic performance, yielding a 20.9% increase in the overall heat transfer coefficient and a 25.8% rise in the comprehensive coefficient of performance compared to the 15° baseline. Consequently, 60° was identified as the optimal cold-side angle. For the hot side, corrugation angles of 22.5° and 35° enhance the overall heat transfer coefficient by 8.5% and 8.8%, respectively, with comprehensive coefficient of performance values exceeding 1.02 times that of the 15° case, indicating an optimal range of 22.5° and 35° . Furthermore, a corrugation pitch of 9 mm and a height of 1.5 mm constituted the optimal geometric configuration. These findings underscore the significant impact of port arrangement on overall performance. Subsequent optimization of the cold-fluid inlet position revealed that, under a constant flow rate, the configuration with an upward-shifted inlet delivered superior overall heat transfer performance.

This study employed a sequential parametric approach to isolate the impact of key corrugation features on thermal-hydraulic performance. While this clearly identifies baseline sensitivities, higher-order interactions among geometric variables warrant further investigation. Future work will couple Response Surface Methodology with multi-objective genetic algorithms to achieve a full multi-parameter optimization.

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Ethics Approval: Not applicable. This study does not involve human subjects or animal experiments.

Conflicts of Interest: The authors declare no conflicts of interest.

Nomenclature

d	Corrugation pitch, m
f	Friction factor
H	Corrugation height, m
h	Convective heat transfer coefficient, W/(m ² ·K)
j	Heat transfer factor
L	Channel length, m
Nu	Nusselt number
Pr	Prandtl Number
ΔP	Pressure drop, Pa
Re	Reynolds number
v	Fluid velocity, m/s
β	Corrugation angle, °
δ	Plate thickness, m
η	Comprehensive coefficient of performance
λ	Coefficient of thermal conductivity, W/(m·K)
μ	Dynamic viscosity, kg/(m·s)
ρ	Density, kg/m ³
h_t	Heat transfer coefficients, W/(m ² ·K)
d_e	The characteristic length of the inter plate channel, m

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